FluiDyne Engineering Corporation

FLOW INITIATION STUDY FOR PROPOSED TUBE WIND TUNNEL PHASE I APPENDIX RE-EVALUATION OF CANDIDATE SYSTEMS

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SUMMARY

The Phase I report (reference 1) on NASA MSFC Contract NAS 8-20214 was submitted by FluiDyne during October 1965. Ratings for candidate flow initiation systems in a large tube wind tunnel were included and a recommendation of a single system for continued development was made. Concurrent studies of utilization and operational characteristics of the proposed tunnel conducted at MSFC have placed renewed emphasis on the importance of interference free flow in the test section and of growth potential for filling in the transonic regime of test capability. This appendix to the Phase 1 report presents a re-evaluation of flow initiation systems with accountability of the evolving utilization specifications and with consideration of possible relaxation of test frequency and operational sophistication in favor of lower initial cost and more certain shakedown to operational status.

1.0 INTRODUCTION

The valve rating system used in reference 1 was based on a range of selection criteria with distributed weighting appropriate to existing specifications. Following submittal of the Phase I report a re-evaluation of valve ratings to accommodate requirements imposed by continuing studies of tunnel utilization was necessary in order to obtain the best current selection of a valve for development. In particular, since many studies in the high Reynolds Number Facility will involve boundary layer transition and turbulence, any valve with an appreciable influence on the free stream turbulence level can not be considered. Thus, in general, valves with residual blockage can not be located upstream of the nozzle. A desired exploration of minimal initial costs for a valve system prompted study of partially destructive systems and their attendant larger operational problems and cost. The certainty of successful development of the candidate valve to a working system is an overriding factor in the selection process for obvious reasons. Thus, systems with questionable certainty on early availability or on reliable cost estimates were severely downgraded, although they might eventually have the greatest potential for a sophisticated test facility.

Elimination of valves having residual blockage from locations upstream of the nozzle means that the most reliable valve concepts can only be used downstream of the test section. Model tunnel tests at MSFC and preliminary flow calculations suggest that there could be a starting loads problem with any downstream valve location. The starting loads problem has been re-examined to make sure that these loads are within reasonable limits for practical, downstream, valve locations.

2.0 ADDITIONAL STUDY WORK

The additional study work carried out in the process of selecting the candidate valve for development included calculations and hydraulic analogy work to determine the influence of downstream valve location on starting time and starting loads. Cost analysis and analysis of development reliability were further amplified.

2.1 Starting Loads

This work was done to help determine which, if any of the valves could be located downstream without increasing the model starting loads to values appreciably greater than the running loads. The work commenced with a rational consideration of the starting process where it was observed that, as the valve location is moved downstream from the nozzle throat, the upstream running expansion which initiates flow in the nozzle becomes more spread out and even though a low initial back pressure exists, the starting process varies from a "quick" start with no notable normal shock wave to a "slow" start with a full strength starting normal shock wave.

Simple calculations were made to determine evacuation time of the test section volume after valve opening to implement a comparison between the evacuation time and the time required for flow initiation (.06 sec). If the time for evacuation was short compared to the flow initiation time, a valid conclusion could be made that no normal shock would develop as long as the initial back pressure downstream of the valve was roughly equal to or lower than the running nozzle exit static pressure (which is the case for Mach numbers through 2.0 at high P_0). If the evacuation time was long compared to the flow initiation time, a strong starting normal shock would occur. Evacuation time depended heavily on the axial position of the valve. The results indicated that for practical valve locations the situation is marginal, that is, the evacuation time and flow initiation time are almost equal, so the occurence of a normal starting shock wave would be likely, although the shock would probably be less than full strength.

These conclusions were confirmed by the hydraulic analogy tests. Any valve location at or downstream of the nozzle exit was associated with a normal starting shock wave. The time required for starting and the strength of the starting shock wave both increased as the simulated valve location was moved downstream but it appeared that a full strength wave would not occur unless the valve location was over three nozzle lengths downstream of the test section. Starting shock strength and starting time for the analogy tests were independent of valve residual blockage as long as the open area was greater than the nozzle exit area.

In conclusion, it appears that the valve could probably be positioned close enough to the nozzle exit so that the starting shock strength would be 1/2 to 1/4 of the full value at the existing stagnation pressure. This, plus the favorable model configuration from the point of view of model loads, lead us to believe that most valves could be designed and positioned axially so that starting loads would not be significantly higher than running loads at the low Mach number, high P_o conditions where a problem might exist.

2.2 Selection of Valves for Re-evaluation

To keep the task of re-rating the valves within reasonable bounds, the number of valve-location combinations considered was limited in a practical way. Only those valves which have no residual blockage were re-rated for the upstream locations. For downstream valves, only the optimum location relative to starting loads, diffuser performance, and adequate open area was considered. The same valve categories apparent in the Phase I report (reference 1) appear here. Two valve subtypes have been eliminated from consideration here, namely: the 2-door normal rotary and the sliding plug. Two new subtypes have been added: a frangible hemisphere, and a sliding sleeve (basically a plug). These valves are portrayed in Figures 1 and 2.

In some cases, the valves need to be supplemented by an upstream tight shutoff valve in order to meet the 90 minutes between runs criteria or to meet some reasonable run schedule. For these cases the cost of the tight shutoff valve is included in the cost of the valve and its inclusion is noted. A list of valve types, sizes, and location combinations which are a part of this new evaluation appears on the following page.

TYPE	SUBTYPE	LOCATION	SIZE	TIGHT SHUTOFF INCLUDED
Gate	Orlin Nozzle	Upstream	5'	No
	2-Door	Upstream	11'	No
	2-Door	Upstream	5'	No
	2-Door	Downstream	6'	Yes
Plug	Sliding Sleeve	Downstream	7'	Yes
Normal Rotary	Multi-vane	Downstream	8'	Yes
	2-Door Collapsible	Upstream	11'	No
	2-Door Collapsible	Upstream	5'	Yes
	2-Door Collapsible	Downstream	6'	Yes
Axial Rotary	18 Port	Downstream	9'	Yes
Frangible	Hemisphere	Downstream	7'	Yes

2.3 Valve Cost Analysis

The valve cost analysis incorporated in the ratings of the Phase I report was limited to the cost level and relative ranking of the candidate valves through design, fabrication, and installation. When consideration of semi-destructive systems such as frangible diaphragms is incorporated into the ratings, a cost comparison including fixed costs per run is mandatory. The operations required to reset the quick-opening valve for a run were visualized for each valve concept and the time and material required were evaluated. No attempt was made to obtain estimates of longer term maintenance costs for this evaluation. The evaluation also assumed that explosive bolts in sufficient quantities to supply full restraint of the quick-release parts under pressure were used on all pertinent systems even though it is recognized that certain of the systems were more conducive to development of re-useable latching methods. The cost of a separate tight shutoff valve was included in the costs of all downstream valves and in the cost of the upstream collapsible valve. The selective inclusion of the tight shut off valve was necessary to avoid making cost comparisons at widely disparate test frequency performance levels for the several valve types rated. The graph on Figure 3 and Table 1 illustrate the results of the cost analysis comparison.

2.4 Estimate of No. of Runs Per Day

Table 2 gives the estimated number of runs per day for each of the valve types. The numbers given are based on: the estimated man hours replacement work necessary on each valve between runs, a reasonable number of workmen in a given work area, and, an eight hour working day. It is of course assumed that any test section work would be completed during the time of replacing and resetting the valve components. The replacement work necessary on each valve is estimated from the number of explosive bolts that must be replaced and reset, the clean-up work required, the frangible elements that have to be replaced, etc.

2.5 Certainty of Successful Design

The design figures and major areas of uncertainties of each of the valves being considered are listed below. Also included are descriptions of the two valves (the frangible disk and the sliding sleeve) which were not included in reference 1.

2.5.1 The Orlin Nozzle

2.5.1.1 Design

The following list gives some the preliminary design figures associated with the 5 ft. orlin nozzle:

Nozzle Weight (each half) (lbs.)	23,000
Initial Pressure Load (lbs.)	1,750,000
Max. Opening Distance (each half) (ft.)	2.5
Max. Acceleration (ft./sec.^{2})	2,450
Max. Velocity (ft./sec.)	75
Kinetic Energy (ft. lbs.)	2,000,000
Length of Seal (ft.) (Primarily Sliding Seal)	42

2.5.1.2 Uncertainties

The principle areas of uncertainty concerning the successful design, fabrication, and operation of the orlin nozzle are listed below.

a. Release Mechanism. The initial pressure loading on the nozzle is quite severe. Explosive bolts may provide one possible method of release, but the large number which would be required, the high replacement costs, and the difficulty of replacement bring their practicality into question.

b. Flexible Plate Structure. The flexible plate thickness was determined from the maximum allowable bending stress (120,000 psi for high strength stainless) and the plate curvature for the closed throat condition. With an unsupported plate the air loads impose additional stresses which are intolerable for current materials. Reference 1 cites several methods which were considered in attempting to reduce the combined bending stresses. Since these did not appear particularly attractive, it seems likely that a considerable amount of work would have to be done in trying to develop a workable concept.

c. Snubber Design. The design of the snubber mechanism for the orlin nozzle presents problems which are similar to the snubber problems in the other valves. Kinetic energy dissipation required is above the range provided by commercially available snubber units. Initial contact velocity is high and thus poses severe problems. Snubber design for the orlin nozzle is additionally complex because total travel is variable (depending on Mach Number), and thus, the snubbers must be designed to assure variable kinetic energies and withstand a wide range of impact velocities. Also, the snubbers must stop the nozzle in precise positions to produce the desired test section Mach Numbers.

d. Seal Design. It is probable that to achieve good test section flow quality the nozzle would have to be sealed at the edges of the moveable plates. This necessitates using sliding seals which pose reliability and maintenance problems. The seal at the throat also poses difficult design problems.

2.5.2 Two Door Gate Valves

2.5.2.1 Design

Below are listed some of the approximate comparative design figures for the various gate valve sizes:

	5'	6'	11'
Gate Weight (each half) (lbs.)	1,300	2,300	12,000
Press. Load on Gate (lbs.)	1,100,000	1,600,000	5,200,000
Edge Brg. Loading (lbs./in.)	20,000	24,000	44,000
Friction Force (u = .1) (lbs.)	110,000	160,000	520,000
Actuator Size (in. dia.)	24	30	76
Actuator Weight (lbs.)	500'	700	4,000
Actuator Force (lbs.)	224,000	384,000	2,720,000
Max. Acceleration (ft./sec.^{2})	2,000	2,400	4,400
Max. Velocity (ft./sec.)	100	120	220
Kinetic Energy (ft. lbs.)	290,000	670,000	12,000,000
Length of Seal (ft.) (Primarily Sliding Seal)	25	30	55

2.5.2.2 Uncertainties

The major areas of uncertainties in the design, fabrication and operation of the 2-door gate valves are listed below. All three sizes of the valves have similar problems; however, the magnitude of the problems varies with the design figures listed in the table above. For example, the snubbing problem is approximately proportional to the kinetic energy, or about 40 times larger for the 11 ft. gate as compared to the 5 ft. gate.

a. Actuator-Snubber System. This system, which is similar to the one on the multivane valve, consists of a preloaded actuator which, when released, pulls the gate or door out of the tunnel flow area within the desired opening time. The door is accelerated until it passes the tunnel wall. The driving force is then relieved and the moving mass is decelerated to a stop by snubbers. One such system is required for each door of the two door gate valves. It is desired that both systems operate simultaneously, thus providing no net external force. To be realistic, however, the systems and their supporting structures must be designed for each one to act individually. This results in very large structural supports, and probably significant shock loads to the tunnel and structure when operation is not simultaneous.

Sizing the actuators and snubbers depends to a large extent on frictional forces on the door and on the piston and rod. These are difficult to predict and probably will vary considerably from run to run, depending on lubrication, wear on the surfaces, temperatures, etc.

The snubber capacities required for the larger valves are beyond the sizes presently designed. Also the velocities at which the snubbers are impacted are well above the present design range.

The 5 ft. valve which would be located upstream of the test section would be designed to reclose after a given run time interval. This sequence adds considerably to the complexity of the control system. Some of the additional problems it raises are:

- Control sequences and times
- Synchronizing the two systems
- Accurate positioning at the closed position
- Time required for seal actuation

b. Bearing Surfaces. The pressure load on the faces of the doors result in very large loads on the supporting bearing surfaces. Using conventional bearing materials results in bearing surfaces several inches wide. As the doors deflect with pressure loading, the contact area on the bearings moves to the inner edges which could cause local yielding and galling. The lubrication of these surfaces is also a problem.

c. Seals. Inflatable seals are required to seal around the edges of the doors. These must deflate in a short time interval, and must retract below the sealing surface so as not to be sheared off or damaged as the door slides. Because of the location, they will be quite difficult to inspect and replace. The static seal at the contacting surface between the doors should be a simpler problem, provided the doors can be positioned accurately when closed.

d. Aerodynamic Contour. This is the problem of providing fairings, or covers to fill the gaps left in tunnel walls when the valve is open. (This would be necessary only in the upstream valve.) These fairings would have to be telescoping or collapsible, yet be able to withstand the air loads.

e. Alignment & Setup. There are difficult problems in fabricating and assembling hardware of this size which must provide a tight seal and must move these distances. The problem of maintaining the initial clearances, straightness, and overall tolerances under the accelerations encountered here appears even harder. Unfortunately, this is the type of problem which cannot be appreciated or measured until after the hardware is built and run.

2.5.3 Sliding Plug Valve

2.5.3.1 Description

The sliding plug valve consists of a 7 ft. dia. x 5 ft. long sleeve which slides axially on a stationary centerbody. The valve is located inside the concrete silencer-exhaust plenum at the exit of the tunnel.

In the closed (upstream) position, the pressure is confined by a static seal at the flange attachment to the tunnel exit and by a dynamic seal contacting the major diameter of the fixed centerbody. The difference in the seal diameters (7 ft. vs. 6 ft.) results in an unbalances force tending to drive the plug downstream. When the flange attachment is released, this force drives the sleeve downstream, thus "opening" the valve. The sleeve movement is stopped by a group of shock absorbers located downstream of the sleeve. A separate hydraulic actuator will return the sleeve to the closed position.

2.5.3.2 Design

Preliminary Design of the sliding plug shows:

Sleeve weight	=	6,600 lbs.
Upstream seal dia.	=	7 ft.
Downstream seal dia.	=	6 ft.
Driving force	=	860,000 lbs.
Max. velocity	=	90 ft./sec.
Max. acceleration	=	130 g's
Shock absorbers	=	6 - 6" Bore x 18" Stroke
Length of seal		22 ft. - Static & 19 ft. - Sliding

2.5.3.3 Uncertainties

The design problems which involve the major uncertainties are:

a. Calculation of Driving Force. As the valve starts to open, the air begins to flow and the driving force diminishes. This plot of force versus opening will determine the acceleration forces, maximum velocity, opening time, and the shock absorber loads.

b. Shock Absorber Design. The kinetic energy involved here is above the design range of the normal shock absorber application. The design of the shock absorber rod, and the "Striker" portion of the sleeve are critical. Also, the possibility of failure of one unit and the resulting "Cocking" loads must be considered.

c. Release Mechanism. The release mechanism must resist at least 860,000 lbs., and must release instantaneously. Although explosion bolts appear feasible, the problems of cost, reliability, ease of installation, safety, etc., must be studied. For example, what happens if one or more of a group of some 24 bolts fails to be exploded?

d. Sleeve Structure. The sleeve must be lightweight, yet strong enough to withstand the pressure, the accelerations, and the concentrated shock absorber reactions.

2.5.4 Multivane Valve

2.5.4.1 Design

Some of the design figures for the 8 ft. x 8 ft. multivane valve are listed on the following page.

Friction Torque	(ft. lbs.)	18,800
Friction Force on Actuator	(lbs.)	45,000
Inertia Force on Actuator	(lbs.)	86,000
Actuator Size	(in. dia)	18
Max. RPM Vanes	(RPM)	600
Max. Rack Acceleration	(ft./sec^2)	1050
Max. Rack Velocity	(ft./sec.)	26
Kinetic Energy	(ft. lbs.)	17,500

2.5.4.2 Uncertainties

The design problems which involve the major uncertainties are:

a. Seals. This valve concept is dependent on elastomeric seals at the junctions:
   - Between the vanes (72 ft.)
   - Between the edge vanes and the walls (16 ft.)
   - Between the ends of the vanes and the walls (16 ft.)

   The seals between the vanes will be static compression type, the remainder will be inflatable. All seals must seal against 600 psi pressure with a total leakage of less than the compressor output. All seals must be attached or confined in slots to prevent being pulled loose by the air flow. In addition, the inflatable seals must retract below the surface when deflated to prevent being sheared off as the vanes rotate. They also must inflate and deflate fast enough to be consistent with valve operation. Other problems which must be investigated are:

- Pressure connection design
- Seal material
- Method of replacement
- Allowable tolerances on vanes and housing; dimensional and surface finish
- Maximum allowable gaps and steps on mating surfaces.

Some of the questions with respect to the static seals have been answered in bench tests. Similar tests for the inflatable seals are planned. Both types must be thoroughly tested in the model valve to be sure of all answers.

b. Bearings. The rotating vanes are supported at several intermediate points as well as at the ends. The bearings at the intermediate points must be small so as to be buried within the vane contour, yet must carry the full pressure loading. Frictional torque must also be kept low to minimize the shaft torques and actuator requirements. Standard needle bearings to carry these large loads are not available. There is a possibility that specials could be made. The next choice is the spherical bushing which has good load carrying capacity, but frictional torque may be too high. Both types of bearings must be tested to measure load carrying capacity and frictional torque. In this same test set-up it will be possible to check out the bearing support design.

c. Actuator-Snubber System. This system must be designed to rotate the vanes 90° in the 1/20 sec. The actuator must be preloaded (probably pneumatically), and released at the desired time. The driving force should be relieved near mid-travel, at which point a snubber (shock absorber) must take over and decelerate the moving parts smoothly to a stop through a given distance. The Preliminary Design of this hardware as shown on Dwg. 0478-901 must be carried into detail design with possibly some developmental testing to solve these problems:
- Time of Actuation
- Snubber Design
- Quick Release Design
- Control Sequence

d. Other Areas. Other design problems which may require considerable effort are:
- Rack & gear tooth design.
- Deflections in housing and stiffener grid.
- Fabrication methods and tolerances for the vanes and housing.

2.5.5 Collapsible Valves

2.5.5.1 Design

Below are listed some of the comparative design figures for the various collapsible valve sizes:

	5 ft.	6 ft.	11 ft.
Door Weight (each) (lbs.)	1,300	2,300	12,000
Press. Load on Door (each) (lbs.)	1,100,000	1,600,000	5,200,000
Max. Snubber Velocity (at impact) (ft./sec.)	300	400	950
Total Energy Per Door (ft. lbs.)	2,100,000	3,600,000	22,400,000
Length of Seal (ft.) (primarily static seal)	25	30	55

2.5.5.2 Uncertainties

The major areas of uncertainty in the design, fabrication, and operation of the collapsible valves are listed below. These problems are common to all the sizes; however, the magnitude of the problem increases greatly with size.

a. Release Mechanism. The holding capacity (size) of the release mechanism is equal to approximately half the pressure loading on each door of the valve. As can be seen from the tabulated valves, these forces are very large, and require many explosion bolts or a large mechanism. If the doors are individually supported, the problem of synchronization arises.

b. Snubbers. The snubbers must be designed to decelerate the doors to a safe stop. The capacities required, based on the kinetic energies listed, are beyond the sizes presently designed. Also, the velocities at which the snubbers are contacted are several times beyone the presently accepted safe values.

c. Seals. Inflatable seals are required to seal around the edges of the doors on all the valves except possible the valve in the downstream location. The inflatable seal problems are similar to those encountered in the other types of valves: deflation time, retraction, gap, confinement and maintenance.

d. Aerodynamic Contour. Any valve located just upstream of the test section would have to present a smooth tunnel wall contour when open. This means that the doors would have to be decelerated and be stopped just flush with the tunnel wall. This requirement restricts the design of the seals, release mechanism and snubbers.

2.5.6 Axial Rotary Valve

2.5.6.1 Design

Some of approximate figures associated with the design of the 9 ft. 18 port downstream axial rotary valve are listed below.

Inertia Torque	(ft. lbs.)	15,400,000
Actuator Force	(lbs.)	2,200,000
Actuator Diameter	(ft.)	5.7
Max. RPM of Rotor		135
Max. Snubbing Velocity	(ft./sec.)	50
Max. Kinetic Energy	(ft. lbs.)	1,500,000
Length of Seal (primarily sliding seal)	(ft.)	230

2.5.6.2 Uncertainties

Major areas of uncertainty regarding the successful design, fabrication and operation of the axial rotary valve are summarized in the following list.

a. Seals. Of all the valves under consideration, the axial rotary has the largest required seal length. This means that seal designs and the associated valve tolerances will be critical factors in a successful valve design.

b. Release-Actuator-Snubber Systems. Like the majority of the valves, the successful design of the axial rotary requires the movement of large masses at high velocities. This in turn requires a large actuator and imposes additional design problems on the release and snubbing mechanism. The problems associated with the release and snubber systems appear to be of the same order of magnitude as the other large valves under consideration.

c. Fabrication and Installation. In fabricating a structure of this size, tolerances in the valve will be critical to maintaining successful operation. Problems of this type are difficult to measure at this stage of development. There has not been a great deal of preliminary design work done on the axial rotary valve because it does not appear to have any particular advantages and does poses some serious disadvantages.

2.5.7 Frangible Disk Valve

2.5.7.1 Description

The frangible disk valve consists of a dished or hemispherical head attached to the downstream end of the tunnel (inside the silencer-exhaust plenum). The disk can be ruptured at a desired time, thus releasing the flow, i.e. "opening the valve". A new disk must be installed for each tunnel run. Two possible methods of rupturing the disk are:

a. Use of primer (explosive) cord laid on the surface of the disk. This cord detonates at the rate of several thousand feet per second and can be sized to cut through various thicknesses of steel. The pattern of attaching the cord to the disk can be adjusted so as to cause the disk to "petal" out upon rupturing, thus minimizing the fragmentation, i.e. amount of shrapnel released.

b. Use of two disks, one in front of the other. Both disks are designed to rupture at less than the tunnel storage pressure. An intermediate pressure will be maintained between the two during pump up. By releasing this intermediate pressure, the upstream disk bursts, releasing pressure to burst the downstream disk, thus "opening the valve".

2.5.7.2 Design

The disks will be 1/2 to 3/4 inches thick steel depending on the alloy. The attachment will be by conventional flanges and bolts. The inside of the plenum will be lined with material to absorb any flying pieces from the disks.

2.5.7.3 Uncertainties

The design problems which involve the major uncertainties are:

a. Disk Design - Explosive Release. A thorough knowledge of explosives must be applied in in the disk design so as to determine:
   - Disk material, thickness and shape
   - Pattern and method of attaching the explosive.
   - Resultant time of opening.
   - Size of the opening.
   - Amount of fragmentation.

   Many of these problems can be answered only through actual tests.

b. Disk Design - Pressure Release. This method of release is more along lines of conventional design, however, design and fabrication tolerances are more critical than with the explosively released disks to insure rupture at given pressure differentials. The following must be determined, probably partly through test:
   - Disk material, thickness and shape.

- Resultant time of opening.
- Size of opening.
- Amount of fragmentation.

c. Design of Holding Flanges. These must be designed for the full pressure area, and also as quickly removed for replacement of the disk(s).

d. Design of Plenum. This must withstand the internal pressure loading as the disks are ruptured. Provisions must be made for absorbing any shrapnel which is released from the disks.

e. Safety Considerations. Special safety procedures must be worked out in the storage, handling, placement and detonation of the explosives.

f. Aerodynamic Effects. Any detrimental effects of the explosive release or the surges due to the two disks rupturing must be studied.

3.0 VALVE RERATING

In Section 3.0 of reference 1, one rating system was presented which included all of the selection criteria. Here, the valves will be rated on a number of bases for the current selection procedure. These include a revised inclusive rating system which will be presented and applied.

3.1 Rating Based on Certainty of Success Only

Section 2.5 of this Appendix contains a discussion of the problem areas associated with each valve type. The review of these problem areas was aimed at providing a basis for ascertaining the probability of success in arriving at a reliable valve design in each case. This certainty or probability must be translated now into a numerical rating N_1 to provide one measure for selecting the valve for development.

It can be seen from Section 2.5 that there are four major problem areas common to all of the valves (with the exception of the frangible disk, in some cases). These problem areas are: the release mechanisms, effective seal designs, the snubbing mechanisms, and the inherent safety of each of the valves. In each of these four areas, it was determined what items actually constitute the major design problems, and which, when objectively rated, could be translated into a relative certainty that the given valve could be developed successfully to give reliable operation.

The major problems in the design of reliable release mechanisms are associated with: the magnitude of the forces on the mechanisms prior to release and the number of explosive bolts which must fire simultaneously to insure proper operation. The problem of designing effective seals involves both the length of seal required and the type of seal (either sliding or compression). The problems in reliable snubber design are associated with the total energy to be dissipated by the snubbers and the velocity at which the valve elements strike the snubbers. Safety problems involve the hazards to personnel in dealing with explosives and in the event of a valve failure and hazards to other equipment due to valve failure, shrapnel, etc.

Each of the eight items listed above, forces on release mechanisms, total energy to be absorbed, velocities, etc. (with the exception of personnel and equipment safety, which were subjective ratings) were calculated for all of the valves and appear in Section 2.5. The relative certainty of success was then determined for each item as follows.

Area of Uncertainty	Estimated Quantity	Factor Desig.	Evaluation
Release Mechanism	Force (F) on Mechanism	a_1	$a_1 = \frac{F_{min}}{F_{ref}}$
	Number of Explosive Bolts (n)	a_2	$a_2 = \frac{n_{min}}{n_{ref}}$
Seals	Seal Length (ℓ)	a_3	$a_3 = \frac{\ell_{min}}{\ell_{ref}}$
	Type of Seal	a_4	$a_4 = 1.0$ if static seals predominate
	$a_4 = 0.5$ if evenly distributed between static & sliding seals		$a_4 = 0$ if sliding seals predominate

Area of Uncertainty	Estimated Quantity	Factor Desig.	Evaluation
Snubbers	Impact Velocity (V)	a_5	$a_5 = 2\frac{V_{min}}{V_{ref}}$
	Energy Absorbed (E)	a_6	$a_6 = \frac{E_{min}}{E_{ref}}$
Safety	Subjective Safety to Personnel	a_7	maximum value 1.0
	Subjective Safety to Equipment	a_8	maximum value 2.0

$$N_1 = a_1 + a_2 + a_3 + a_4 + a_5 + a_6 + a_7 + a_8$$

The sum of the eight ratings (a_N) equals the overall certainty of success rating (N_1) as shown.

In some cases, $Item_{min}$ was zero (for example, the energy to be absorbed by the snubber in the frangible disk). In these cases, the valve with $Item_{min} = 0$ was given the maximum points for that rating and the valve with the next lowest magnitude for the item was used as $Item_{min}$. The safety ratings were based on knowledge of the magnitude of the forces, energies, direction of valve blockage motion, etc.

3.2 Ratings Based on Cost Only

The initial cost and operating cost for each valve are discussed in Section 2.3 and presented in Table 1 and Figure 3. Since cost by itself may play an important part in determining which valve should be developed, we will define two ragins N_2 and N_3 which are based entirely on the estimated cost of each valve. N_2 will be based on the total initial cost while N_3 will be based on the total initial cost plus operating costs for 100 runs. Their values will be calculated as follows:

$$N = 10 \frac{C_{max} - C_{ref}}{C_{max} - C_{min}}$$

where C is cost in dollars.

The resulting rating values appear in Table 2 along with N_1 and N_4 (which will be defined below in 3.3).

3.3 Inclusive Rating

In many respects the inclusive rating developed here is similar to that presented in Section 3.1 of reference 1. The rating value, N_4 , will be formed from a number of multipliers as follows:

$$N_4 = M_1 \times M_2 \times M_3 \times M_4 \times M_5 \quad .$$

M_1 , as before, measures the basic feasibility of the particular valve-location combination in terms of whether or not it permits the attainment of the tunnel performance specifications. M_1 is also made up of a number of multipliers:

$$M_1 = m_1 \times m_2 \times m_3 \times m_4 \quad .$$

The multipliers m_2 , m_3 and m_4 are defined and evaluated exactly the same as they were in reference 1. Multiplier m_1 has the same definition as before but its evaluation has been changed somewhat in that it will have a value of 0.8 rather than 0.5 when there is a possibility of instrumentation or model structural problems arising because the test chamber must be at full storage tube pressure prior to a run. Otherwise, its evaluation will remain the same. This change only affects the ratings of the downstream located valves, and it was made partly because of comments by Marshall Space Flight Center personnel which indicated that this might not be as serious a condition as had originally been supposed.

The M_2 of reference 1 has been eliminated (prerun test chamber pumpdown doesn't reduce starting loads significantly for the low Mach number, high P_0 conditions where a problem might exist). M_2 will herein be defined the same as M_3 in reference 1. That is, it measures the existence of any hazard to either the valve or model due to the other. If there is no hazard $M_2 = 1.0$ while $M_2 = 0.75$ if a hazard exists.

The present M_3 measures the cost of the valve and whether or not the valve can be closed mechanically at pressure. It does not contain any measure of the certainty of successful development as did M_4 in reference 1. M_3 is evaluated as follows:

$$M_3 = N_3 + A_1$$

where N_3 is the cost rating developed in the preceding section and A_1 is equal to 5.0 if the valve can be reclosed mechanically against pressure (zero, if not).

Certainty of successful development was deemed an important enough factor to be included as a multiplier in this new rating rather than as one in a series of additive factors. M_4 which measures it is simply equal to $N_{1/10}$.

The inclusive rating presented in reference 1 contained no measure of whether or not the valve could meet the 90 minute between run requirement. In the current rating M_5 measures this where

$$M_5 = \frac{\text{no. of runs per day}}{5} .$$

In conclusion, we have provided an inclusive rating here which differs from the original rating in its greater emphasis on certainty of successful development and in its inclusion of run frequency as a factor in evaluation.

4.0 DISCUSSION OF RESULTS

Table 2 contains a summary of the ratings developed using the rating methods discussed in Section 3.0 and the performance and cost information covered in Section 2.0. These ratings will now be applied as a guide in selecting the most promising valve system for development.

The first rating N_1 is a measure of the certainty that the particular valve can be developed successfully. Both the multiple butterfly and the frangible hemisphere rate high here. The frangible hemisphere, which one expects to be destroyed each run, has few reliability problems as does the multiple butterfly. One finds the next valves in line quite a bit lower on the rating scale. These are the sliding sleeve plug valve and the 5' upstream gate valve. The forces, inertias, and velocities associated with these two valves are considerably higher than those associated with the multiple butterfly.

N_2 and N_3 are both related to cost. On the basis of initial cost only (N_2) the frangible hemisphere rates highest with the sliding sleeve, multiple butterfly, and 5' upstream gate valves running close behind in the rating. When the operating costs are included (N_3), these same four valves are at the top except that the multiple butterfly rates highest. Design and development costs on these valves are also lower than on the other valves with the frangible hemisphere being by far the least expensive to design and develop.

The inclusive rating (N_4) places the multiple butterfly valve first with the 5' upstream gate second and the frangible hemisphere and sliding sleeve third and fourth, respectively. Low runs per day brings the rating of the frangible hemisphere down in this case. Uncertainties lower the rating of the gate valve and the slidin g sleeve. The multiple butterfly, 5' upstream gate, and sliding sleeve valve are the only ones which come close to meeting the required time between runs criterion.

In all of the ratings, the multiple butterfly, 5' upstream gate, sliding sleeve, and frangible hemisphere valves rate consistently higher than the other valves so it seems clear that our problem of selection has narrowed down to these four valves. Table 3 contains more detailed cost estimate breakdowns for these valves.

Except forthe uncertainties involves in development, the 5' upstream gate valve would be highest in the inclusive rating results. It avoids problems associated with downstream valves and can be reclosed against pressure. Nevertheless, we feel that the uncertainties warrant the somewhat lower rating it has been given.

If certainty of successful valve development were the sole criterion for valve selection, the choice would be between the frangible hemisphere and the multiple butterfly. As far as costs are concerned, there are no significant differences between the four top valves although the frangible hemisphere rates highest on the basis of initial cost while the multiple butterfly rates highest when operating costs are included. The multiple butterfly rates a strong first place in the inclusive rating with the gate valve a good second. From these observations, it is our conclusion that development work should proceed on the multiple butterfly valve. Although there may be some argument that the frangible hemisphere is more likely to be successfully developed and therefore, with its lower initial cost be most suitable, we feel that development of the multiple butterfly offers the most in terms of potential achievement partly because it will result in a valve having a good deal of value in applications beyond the immediate one.

During the rerating, the importance of including the entire transonic regime through Mach number 1.0 in the range of operation was kept in mind. The recommended valve configuration will not limit the range of transonic operation. Changes in tunnel size will not influence the valve selection either. Most of the problems will scale up with tunnel size.

A problem which was not considered during the rerating but which ought to be mentioned involves axial loads on the tunnel during valve operation and consequent foundation problems. With valves which discharge axially there is a sudden change in the axial force at the valve location when the valve is opened. This can amount to several million pounds. With valves which open axially there may be snubbing loads approaching two million pounds. Consequently, it may be desirable to consider a valve which discharges radially and which produces no axial loads during snubbing. This could be accomplished by a variation of the sliding sleeve valve which utilized two opposing sleeves travelling in opposite directions and balancing out forces rather than one sleeve with its unidirectional snubbing load. Such a valve would be a bit more complicated than the sleeve valve that was rated, but snubbing loads would be reduced some, placing its cost and reliability not far from the single sleeve version.

5.0 REDEFINITION OF PHASE II TASKS

Some preliminary development work has already been done on the multiple butterfly components as a result of the original Phase I rating. On the basis of the rerating, the development work and verification of design concepts of the multiple butterfly valves will be continued through the remainder of Phase II. The following section outlines the Phase II tasks that have been performed or are planned and which enable us to proceed to the final design of the multiple butterfly valve.

The preliminary design and testing of the multiple butterfly valve will be concentrated on the following problem areas:

a. seals
b. bearings
c. actuator and control system
d. snubber .

Preliminary seal test rigs have been built to test the three principle types of seals: the intermediate static seals between the vanes, the seals at the ends of the vanes, and the seals between the outer vanes and the valve housing. The intermediate seals have been tested, and the seal geometry, material and durometer have been determined. Inflatable seals have been tested for use as a seal along the ends of the vanes and between the outer vanes and the valve housing. The inflatable seals appears quite satisfactory for these applications.

A bearing test rig has been designed for testing both needle bearings and ball bushings. In this test rig, the effects of shaft deflections on bearing operation, due to simulated design air pressure loads, can be determined. Also, bearing life and frictional torque can be estimated in this test rig.

A 20" x 20" working section of the valve has been designed which will be tested at full operating conditions. It will contain two vanes, each full size in cross section and 20" long. Each vane will have three intermediate bearing supports and the stiffener gridwork will be identical to that anticipated in the full scale valve. The seals, bearings, and shafts will also be full scale. The face width of the gears and rack, the actuator and the snubber will be scaled down to match the inertia and friction forces of the model. The valve will be tested at full design conditions and the following items will be investigated:

a. Seals - The model valve will provide a further check on seal effectiveness and will be used to verify the results from the seal test rigs. Also, anticipated leakage and required tolerances of the full size valve will be estimated.

b. Bearings - The model valve will also provide a further check on bearing operation and friction with actual vane deflections and operating conditions.

c. Actuator and Control System - A time history of the inertia and friction forces, which are encountered during opening under design conditions, will be determined. The effectiveness of the actuator and release mechanisms will be checked.

Also, the effectiveness of such items as the explosive bolt shroud, rack guides, vane torsional strength, etc., can also be checked.

d. Snubber - A time history of the snubber decelerating forces will be determined. Also, recovery time, overtravel, rod and rack damage due to impact, and the other snubbing problems will be investigated.

The knowledge gained from these tests will enable us to proceed to the final design with a high degree of certainty that the valve can be built and operated successfully.

6.0 REFERENCES

1. Grunnet, Kamis, Thornberg, Hasselquist, and Hamre, "Flow Initiation Study for Proposed Tube Wind Tunnel, Phase I, Preliminary Evaluation of Candidate Systems', Phase I Report to NASA Marshall Space Flight Center Under Contract No. NAS 8-20214, October 1965.

TABLE 1. VALVE COST SUMMARY

Valve Type	Total Initial Cost	Replacement Cost/Run	No. of Runs/Day
Upstream, 11 ft., 2 dr., Gate	$1,600,000	$4,000	1
Upstream, 5 ft., 2 dr., Gate	$ 365,500	$ 800	4
Upstream, 11 ft., 2 dr. Collapsible	$1,600,000	$8,000	.5
Upstream, 5 ft., 2 dr. Collapsible w/T.S.O.	$ 530,000	$2,000	2
Orlin Nozzle	$1,300,000	$5,000	1
Downstream, 6 ft., 2 dr. Gate w/T.S.O.	$ 630,000	$1,100	3
Downstream, 6 ft., 2 dr. Collapsible w/T.S.O.	$ 630,000	$3,000	2
Downstream, 9 ft., 18 Port Axial Rotary w/T.S.O.	$1,330,000	$2,000	1
Downstream, 8 ft., Multiple Butterfly w/T.S.O.	$ 392,700	$ 100	5
Downstream, 7 ft., Sliding Sleeve Plug w/T.S.O.	$ 329,000	$1,200	4
Downstream, 7 ft., Frangible Disk w/T.S.O.	$ 197,600	$4,000	2

TABLE 2. RESULTS OF INDIVIDUAL VALVE RATINGS

Valve Type and Location	N_1	N_2	N_3	N_4
Upstream, 11 ft., 2 dr., Gate	0.56	0	2.01	0.07
Upstream, 5 ft., 2 dr., Gate	2.96	8.80	9.80	3.46
Upstream, 11 ft. 2 dr. Collapsible	0.94	0	0	0
Upstream, 5 ft., 2 dr. Collapsible w/T.S.O.	2.51	7.63	8.37	0.83
Upstream, 5 ft. Orlin Nozzle	1.04	2.14	3.01	0.03
Downstream, 6 ft., 2 dr. Gate w/T.S.O.	2.65	6.91	8.33	1.04
Downstream, 6 ft., 2 dr. Collapsible w/T.S.O.	2.33	6.91	7.37	0.54
Downstream, 9 ft. 18 Port Axial Rotary w/T.S.O.	1.38	1.92	4.36	0.14
Downstream, 8 ft. Multiple Butterfly w/T.S.O.	9.22	8.60	10	5.53
Downstream, 7 ft. Sliding Sleeve Plug w/T.S.O.	3.27	9.06	9.79	2.05
Downstream, 7 ft., Frangible Disk w/T.S.O.	8.00	10	9.04	2.29

TABLE 3

COST BREAKDOWN ON FOUR BEST VALVES

Item	5 ft., 2 dr. Gate Valve	8' Multi-Vane Valve	7' Sliding Plug Valve	7' Frangible Disk
Development and Testing	$ 45,600	$ 57,000	$ 26,000	$ 19,500
Final Design	$ 28,000	$ 24,700	$ 23,400	$ 6,500
Fabrication and Engineering Follow-Up	$245,000	$142,000	$125,000	$ 32,000
Assembly Installation and Checkout	$ 38,900	$ 32,500	$ 19,500	$ 6,500
Other	$ 8,000	$ 6,500	$ 5,100	$ 3,100
Tight Shut-Off Valve	---	$130,000	$130,000	$130,000
Total	$365,500	$392,700	$329,000	$197,600

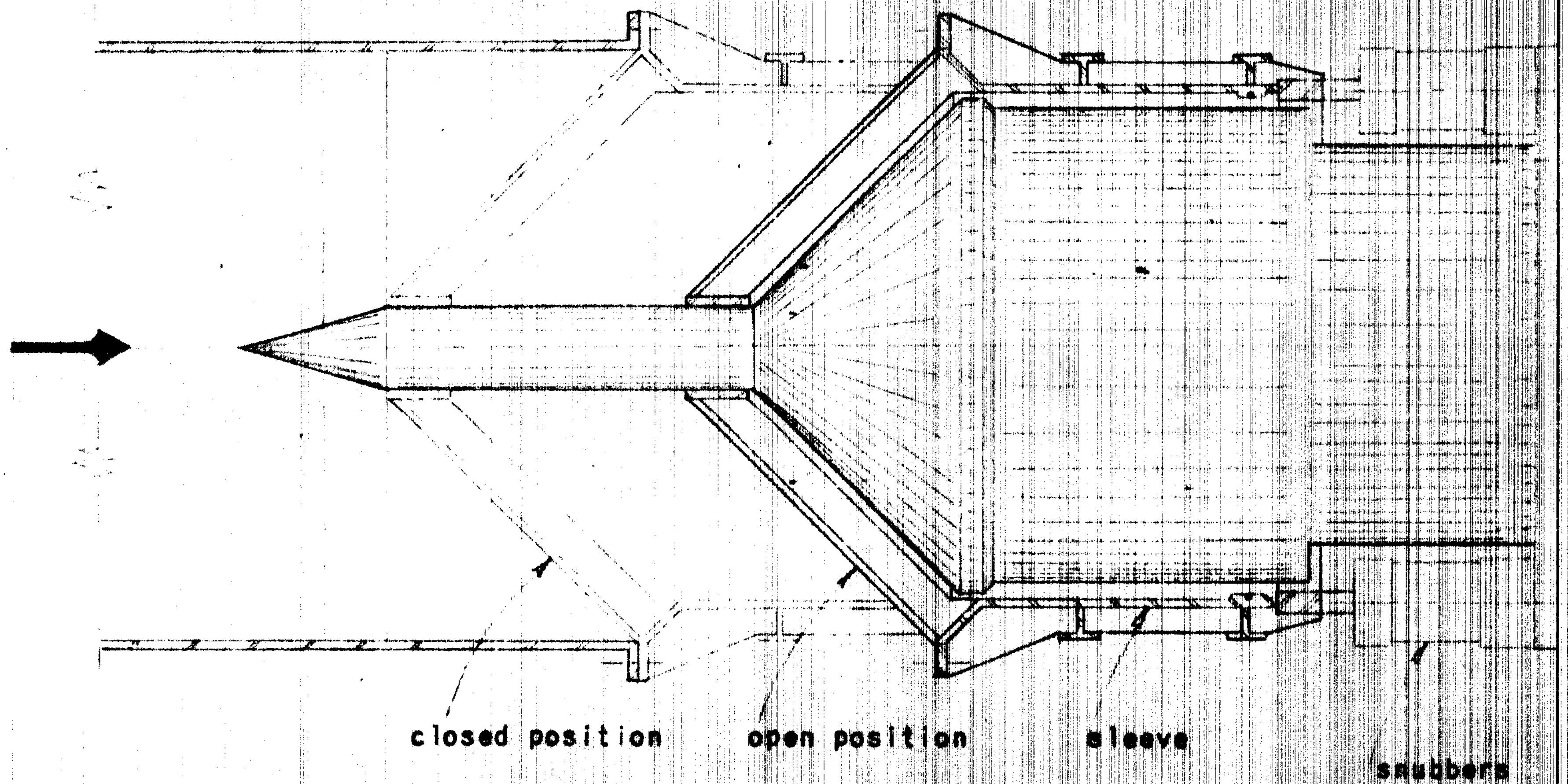


FIGURE 1 SLIDING SLEEVE VALVE CONCEPT

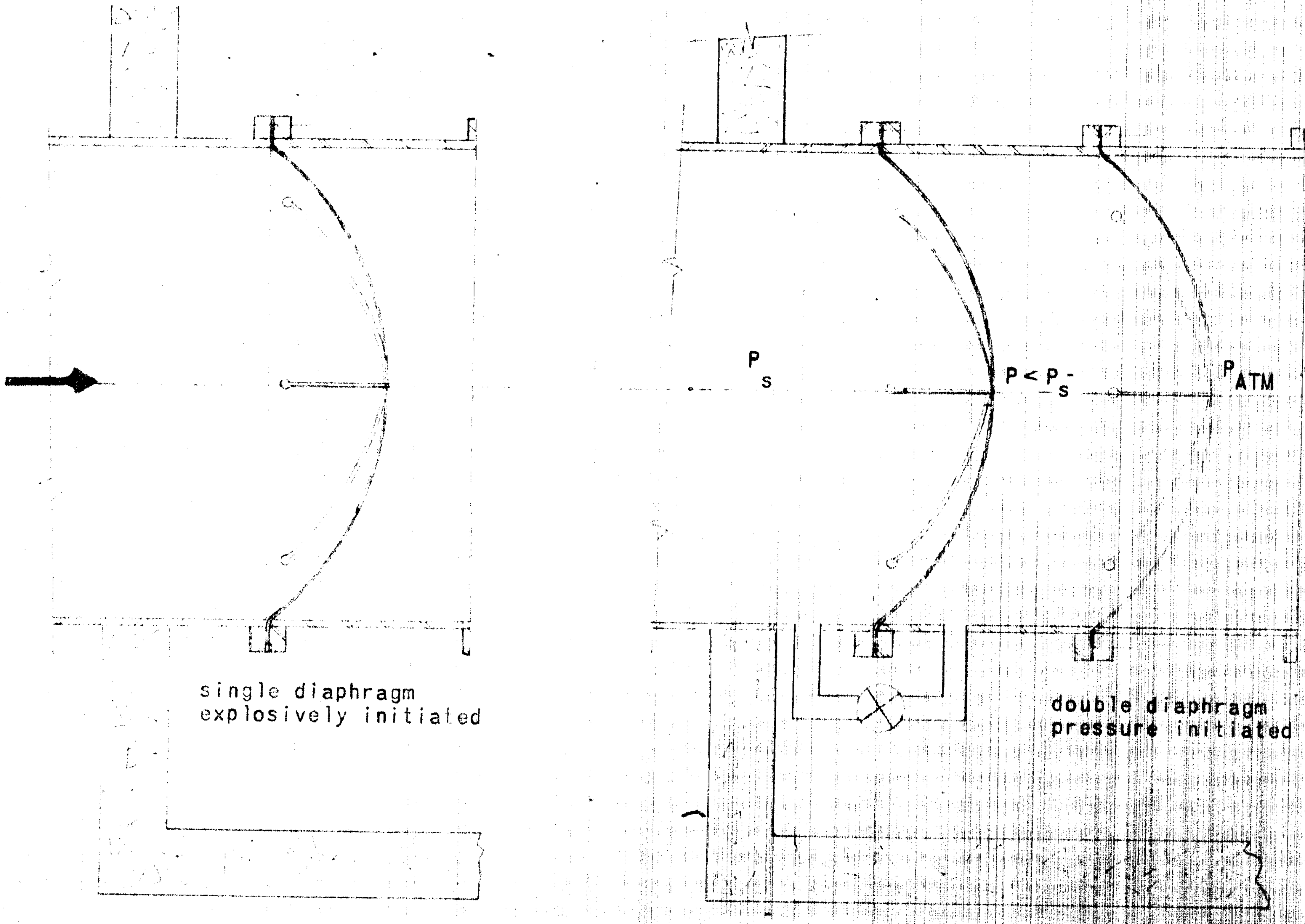
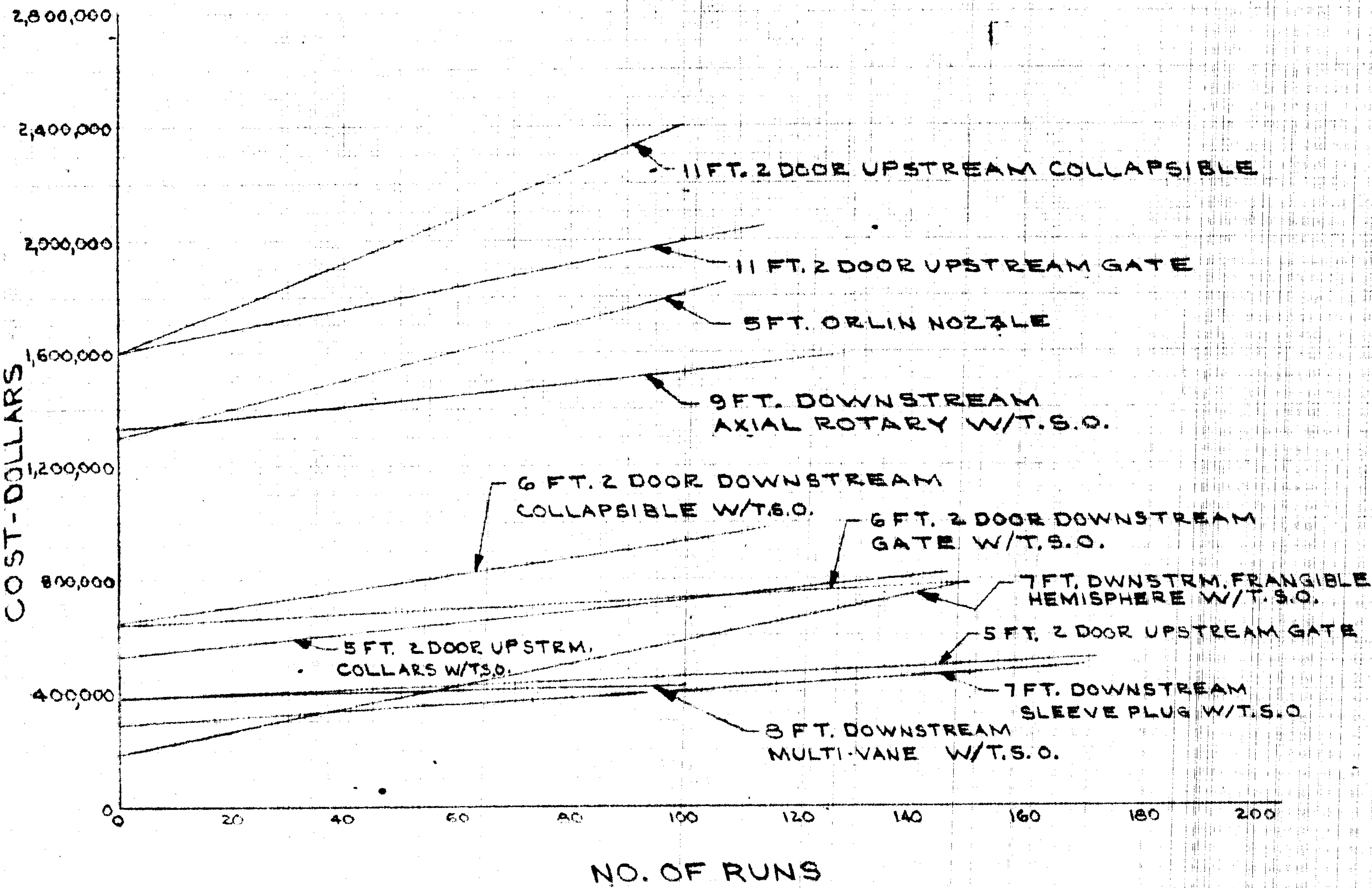


FIGURE 2 FRANGIBLE VALVE CONCEPT

FIGURE 3 TOTAL COST VS NUMBER OF RUNS



FLUIDYNE ENGINEERING CORPORATION

JOB ______ CODE ______ SHEET NO. ______ OF ______

DATE ______

BY ______

ITEMIZED FABRICATION COST [illegible]

5 FT DIA. [illegible]

NO. REQD.	ITEM	WEIGHT (LBS)	PRICE/LB	COST
2	GATES	1100 LB	$3.70	$8570
2	PISTON RODS	110	5.20	1150
2	PISTONS	250	10.90	2730
4	SIDE WALLS	9100	1.65	[illegible]
2	TOP & BOTT. FRAME	1830	1.63	5,910
4	TOP & BOTT. WALLS	2530	1.63	16,[illegible]
2	PRESS. BOX RIGS.	4380	1.30	11,250
2	SNUBBER FRAME	5072	1.63	16,500
2	AIR STORAGE [illegible]	1400	1.30	3,640
2	CYLINDERS	1805	2.08	7,500
2	CONNECTOR SLEEVE	312	1.50	975
4	SIDE BEARINGS	705 (TOTAL)	3.90	2750
2	BOTTOM BEARINGS	350 (TOTAL)	3.90	1,370
	SEALS			3900
	SUPPORTS			3,900
	SNUBBERS			30,000
	VALVE CONTROLS	10,000 (APPROX)		26,000
	PRESS. RETAIN. CYLINDERS			2,600
	SWISS			1,3[illegible]
	MISC.			22,800
TOTAL		90,130	2.78	228,37[illegible]

FLUIDYNE ENGINEERING CORPORATION

JOB ______________________ CODE ________ SHEET NO. ______ OF ______

DATE ______________

BY ______________

ITEMIZED FABRICATION COST BREAKDOWN
8 FT INLET VANE [illegible]

NO. REQ'D	ITEM	WEIGHT (EA.)	PRICE/LB	COST
1	HOUSING & INTERNAL STRUTS	36,000 LB	1.16	41,500
10	VANES	415	6.27	26,000
10	SHAFTS	22	0.71	1,560
100	BRACKETS	30	1.82	5,470
200	BEARINGS			1,170
10	BEARING RINGS			1,040
10	SPLIT RINGS			780
10	[illegible]			1,170
10	SPACERS			390
10	GEARS			1,300
1	RACK	15,321 (APPROX.)		1,040
1 SET	RACK GUIDES			2,600
1	ACTUATOR			6,500
1	SNUBBER			3,900
	SUPPORTS			2,600
	SEALS			6,500
	[illegible]			4,000
	[illegible]			13,000
TOTAL		55,370	2.27	125,950

FluiDyne Engineering Corporation

JOB ______________________ CODE ________ SHEET NO. ____ OF ____

DATE ____________

BY ____________

ITEMIZED FABRICATION COST [illegible]

7 FT. SLIDING SLEEVE VALVE

NO. REQD	ITEM	WEIGHT (EACH)	PRICE/LB	COST
1	MOVABLE SLEEVE	6200 LBS	2.60	16,120
1	CENTER BODY	12,000	1.30	15,600
1	EXIT PIPE- 4'ø: [illegible]	5,300	1.30	6,890
1	EXIT FLANGE	2,000	1.30	2,600
1	FLANGE & MTR. EXT. FRAME	10,000	1.30	13,000
1	UPSTREAM SPIDER			2,500
1	BEARING RING UPSTRM. END			850
1	BEARING SHOES UPSTRM END			1,200
1	RELEASE MECH. (BOLTS, ETC.)			1,300
6	SHUTTERS	10,000 (APPROX)		35,000
1	ACTUATING SYS.			650
	CONTROLS & INTERLOCKS			6,500
	MISCELLANEOUS (SEALS, ETC.)			7,540
TOTAL		55,500	1.98	109,750

FluiDyne Engineering Corporation

JOB ______ CODE ______ SHEET NO. ____ OF ____

DATE ______

BY ______

ITEMIZED FABRICATION COST BREAKDOWN

FOR [illegible] DISK

NO. REQD.	ITEM	WEIGHT (EACH)	$/LB	COST
1	[illegible] HEAD	1,700 LBS	1.15	1,950
2	FLANGE	1630	3.19	10,400
1	[illegible] STRAIN [illegible] TIE & STRAKES	4100	0.32	1,300
	[illegible] EQUIP.			[illegible]
	[illegible]			250
	[illegible]			[illegible]
	BOLTS	10,000 (APPROX)		1,950
	FREIGHT CHARGES MATERIAL			250
	PLENUM CHAMBER STIFF [illegible]			4,000
	MISCELLANEOUS			3,120
TOTAL		19,060	1.51	28,800

ITEMIZED FABRICATION COST BREAKDOWN

TIGHT SHUT-OFF [illegible]

(SLOW ACTING SET, 13 FT. [illegible])

NO. REQ'D	ITEM	WEIGHT (EA.)	PRICE/LB	COST
1	GATE	2200	3.9[illegible]	8,6[illegible]
2	SIDEWALLS	9100	1.65	[illegible]
2	TOP & BOTTOM FRAME	1200	1.6[illegible]	3,900
3	WALLS	2500	1.6[illegible]	12,[illegible]
1	FLANGE	4400	[illegible]	5,700
	BOTTOM BEARINGS			1,300
	SEALS			[illegible]900
	SUPPORTS	5,000 (APPROX.)		1950
	ACTUATOR			2600
	MISCELLANEOUS			2650
TOTAL		39,700	1.83	72,500

PRELIM. DESIGN & DEVELOP.	10,000
FINAL DESIGN	30,000
FABRICATION	72,500
INSTALLATION	17,500
TOTAL INITIAL COST	$130,000